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RESEARCH MEMORANDUM

EFFECT OF BLADE-TIP CROSSOVER PASSAGES ON NATURAL-CONVECTION WATER-COOLING OF GAS-TURBINE BLADES

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NATIONAL ADVISORY COMMITTEE
FOR AERONAUTICS
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EFFECT OF BLADE-TIP CROSSOVER PASSAGES ON NATURAL-CONVECTION WATER-COOLING OF GAS-TURBINE BLADES

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SUMMARY

A water-cooled turbine was fabricated and tested to determine the effect of a connecting passage at the turbine rotor blade tip between a radial coolant passage 0.10 inch in diameter (length-diameter ratio = 25.5) and radial coolant passages in the length-diameter range of 5.1 to 20.4. Coolant flow through the connecting passage is induced by free-convection forces in the radial passages.

A theoretical relation for heat transfer by natural convection from flat plates and in large-diameter tubes (small length-diameter ratio) was adequate to describe the heat transfer in the coolant passages of the turbine of this investigation (length-diameter range of 5.1 to 25.5). Calculated blade-metal temperatures that accounted for the heat-conduction process in the blade metal agree satisfactorily with measured temperatures. Since the blade-to-coolant heat-transfer coefficients are high with water as coolant, the blade temperature is largely determined by the coolant temperature.

The cooling effectiveness of the configuration of this report is slightly greater than that obtained when the radial passages are not connected at the blade tip. Passages of length-diameter ratio of 25.5 were cooled more effectively when connected by a crossover passage at the blade tip to a passage of length-diameter ratio of 5.1 than when connected to a passage of length-diameter ratio of 20.4. The maximum temperature reduction (about 25° F), as reflected by the improved effectiveness, is probably slight for most practical applications.

INTRODUCTION

Analytical investigations such as that reported in reference 1 indicate that the natural-convection heat transfer in small coolant passages in a turbine rotor blade can be improved by providing a crossover passage at the blade tip that connects the small-diameter passage with larger-diameter passages. The present investigation was conducted at the NACA Lewis laboratory to obtain experimental information on this effect and to determine the size of the larger-diameter passage required to furnish the coolant flow for a small-diameter passage without impairing the cooling effectiveness in the larger passage. This investigation was conducted in conjunction with the investigation of reference 2, in which the effect of closed-end coolant-passage diameter on natural-convection heat-transfer characteristics was determined for a range of hole sizes from 0.10- to 0.50-inch diameter.

As discussed in references 1 and 2, natural-convection circulation 3952
of a coolant in closed-end coolant passages requires that there be flow in opposite directions within the blade passages. In small passages the heated boundary layer can occupy such a large proportion of the passage flow area that natural-convection circulation is hampered. Some evidence of the interference of the coolant-flow streams is indicated in reference 2 by the higher coolant temperature at the blade tip for the smaller-diameter passages than for the larger passages. Reference 1 suggests that this interference could be overcome if the coolant in the small hole were supplied at the blade tip. The circulation and cooling effect would be considerably improved. The flow in the small-diameter passages could be generated by the free-convection forces associated with the coolant temperature rise and would be in a single direction, radially inward. In this way the cooling effectiveness could be improved for small passages (large length-diameter ratios) even above that for pure natural-convection circulation in large passages.

The turbine described in reference 2 was altered to provide cross-over passages at the blade tips connecting the 0.10-inch-diameter trailing-edge passage with leading-edge passages ranging from 0.125 to 0.50 inch in diameter. The results of the tests conducted on this turbine with water as the coolant are presented herein for the following conditions:

Coolant-passage length, in. 2.55
Coolant-passage diameter, in.
  Trailing edge 0.10
  Leading edge . . . . 0.125, 0.25, 0.375, and 0.50
Turbine-inlet temperature, °F . . . . 600 to 1200
Turbine rotor speed, rpm . . 4,000 to 12,000, corresponding to turbine tip speeds of 245 to 733 ft/sec and product of Grashof and Prandtl numbers range of order 10^{12}
Coolant inlet temperature, °F . . . . 78 to 96

The temperature data were used to obtained heat-transfer correlations for comparison with theory, to compare with analytically predicted blade-metal temperatures, and to determine cooling effectiveness for comparison with the cooling effectiveness of blades with closed-end coolant passages.

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SYMBOLS

The following symbols are used in this report:

A	cross-sectional area of coolant passage, sq ft
b	length x to spanwise location of blade thermocouples, ft
c_p	specific heat of water at constant pressure, assumed = 1.0 Btu/(lb)(°F)
d	coolant-passage diameter, ft
Gr	Grashof number, $\dfrac{\omega^2 r\beta(T_w - T_5)l^3}{\nu_f^{\,2}}$
h	blade-to-coolant heat-transfer coefficient, Btu/(sq ft)(°F)(sec)
k	thermal conductivity of water, Btu/(ft)(°F)(sec)
l	total length of coolant passage in blade and rim, ft
Nu	Nusselt number, hl/k_f
Pr	Prandtl number, $c_p\mu_f/k_f$
$Q_{b,t}$	total heat flow from gas to blade, Btu/sec
Re	Reynolds number, $w_c d/A\mu_f$; $w_c b/A\mu_{av}$
r	radius to blade midspan, ft
T	temperature, °F
$T_{g,e}$	effective gas temperature for gas-to-blade heat transfer, °F
w_c	coolant flow through leading-edge passage, lb/sec
$w_{c,R}$	coolant flow into rotor, lb/sec
x	passage length from tip end of blade, ft
β	coefficient of thermal expansion of coolant, 1/°F
μ	absolute viscosity of coolant, lb/(sec)(ft)

ν kinematic viscosity of coolant, sq ft/sec

Φ temperature-difference ratio, $\dfrac{T_{g,e} - T_b}{T_{g,e} - T_{in}}$

ω angular velocity, radians/sec

Subscripts:

av average coolant temperature, $(T_5 + T_6)/2$, $^{\circ}F$ 3952

b blade

d based on coolant-passage diameter

f based on film temperature, $(T_w + T_5)/2$, $^{\circ}F$

in coolant at rotor inlet

le leading-edge region of blade

te trailing-edge region of blade

w coolant-passage surface

x based on length x

1,2,3
4,5,6 thermocouple locations (fig. 2)

APPARATUS

The apparatus is the same as that described in reference 2, except that crossover holes were drilled at the blade tip to connect the larger leading-edge coolant passage with the smaller trailing-edge coolant passage as shown in figure 1. The turbine wheel was divided into quadrants of five blades each. The blades of each quadrant had coolant passages of the same configuration, but the diameter of the leading-edge passage varied from quadrant to quadrant. Leading-edge holes were of the following diameters: 0.125, 0.25, 0.375, and 0.50 inch. The diameter of the trailing-edge passages was 0.10 inch. Hereinafter, the various quadrants are designated 0.125-inch quadrant, 0.25-inch quadrant, and so forth. The coolant to each quadrant was kept separated in the turbine wheel to permit temperature-rise measurements for calculating the heat rejection from blades in each of the quadrants.

The instrumentation for the turbine was the same as described in reference 2. The turbine-blade thermocouple locations are shown in figure 2. This instrumentation was the same in each of the quadrants with the exception of thermocouple 4, which was installed only in the 0.125- and 0.50-inch quadrants to obtain representative extreme temperatures.

PROCEDURE

Experimental Procedure

The experimental procedure for this investigation was exactly the same as that described in reference 2. During a typical run, constant turbine wheel speed and essentially constant combustion-air flow and gas temperature were maintained. A range of coolant flows was supplied to the turbine, and pertinent operating data were recorded at each weight flow. Table I summarizes the range of turbine operating data reported herein.

Calculation Procedure

The calculation procedure for this investigation, insofar as applicable, is discussed in reference 2. In brief, these procedures are as follows: Rotor-to-coolant heat flows were calculated from the measured coolant temperature rise and the coolant flow into the rotor. The heat transferred to the coolant through the rotor rim, including the heat conducted from the blade to the rim, was subtracted from the total in order to obtain the blade-to-coolant heat flow. The rim-to-coolant heat flow was that conducted through the rim because of the temperature difference as determined by the measurements at the outer and inner rim (fig. 1). This heat flow and the coolant-passage internal surface area, neglecting the surface area of the crossover passage, were used to calculate the blade-to-coolant heat-transfer coefficient where the temperature difference for heat transfer is the difference between the coolant-passage-surface temperature and the coolant temperature T_5 (fig. 2). Coolant-passage-surface temperatures were calculated from the measured temperatures near the passages by taking into account the heat transfer by conduction within the blade metal. An electric analog was employed to determine the temperature difference between the coolant-passage surface and the thermocouple location. Temperature differences were not calculated for the 0.375-inch quadrant, because, as mentioned in reference 2, no analog was constructed. The experimental blade-to-coolant coefficients are thus obtained and are compared with theory on the basis of the nondimensional parameters, Nusselt number and the product of Grashof and Prandtl numbers.

As a further attempt to evaluate the agreement between the measured temperatures and those predicted from theory, blade temperatures were calculated for comparison with the experimentally measured values. The choice of the theoretical relation used to predict blade-to-coolant coefficients for this calculation is discussed later. The theoretically predicted coefficients and the experimentally measured blade-to-coolant heat flows and coolant temperatures T_5 enabled calculation of coolant-
passage-surface temperatures for a given turbine speed. The blade-metal 3952
temperatures at stations 1 and 3 (see fig. 2) were then determined with the aid of the electric-analog results.

Determination of Heat-Transfer Relation for Blade-Temperature Calculation

Trailing-edge passage. - The fluid temperature in a coolant passage depends on the heat flow per unit volume of coolant and the degree of interference between the heated boundary layer and the central core. If it is assumed that these conditions cause the coolant temperature in the trailing-edge passage to be higher than that in the larger leading-edge passage, a tendency for coolant flow from the leading-edge passage into the trailing-edge passage would exist. Under these conditions the relationship developed in reference 1 for the loop-circuit type of flow could be applied to the heat flow within the trailing-edge passage. The Nusselt number Nu at a given product of Grashof and Prandtl numbers GrPr is shown in reference 1 for the case of $l/d = 25$, which is approximately that of the trailing-edge passages of this turbine. In reference 1 the loop-circuit flow is compared with the flow in blind passages of $l/d < 5$. The Nusselt number for loop flow is only about 30 percent greater than for flow in the blind passages. As shown in reference 2, variations of ±50 percent in Nusselt number have only a small effect on blade temperature (±5° F). In the range of this investigation the average variation of about 5° F is well within the experimental accuracy. Therefore, the prediction of the heat-transfer coefficients for blade-temperature calculations was based on the theoretical relation for turbulent natural-convection heat transfer from flat plates and in tubes of large diameter ($l/d < 5$). The equation as developed in reference 1 is

$$Nu = 0.0210(GrPr)^{0.4} \tag{1}$$

where the characteristic dimension is l.

Leading-edge passages. - With the assumed flow pattern discussed, the condition of counterflow would exist within the leading-edge passages; that is, a portion of the coolant entering the passage forms the heated boundary layer and tends to be circulated radially inward adjacent to the coolant-passage walls. The remaining portion of the coolant, the through-flow, entering the passage would then be circulated through the trailing-edge passage. In order to establish the range of counterflow conditions in the leading-edge passages, the through-flow rate was determined by the Reynolds numbers calculated for the trailing-edge passage by the relation developed in reference 1:

$$Re_{d,te,f} = \frac{0.531 Gr_{te}^{0.513}}{Pr_{te}^{0.308}} \left(\frac{d}{l}\right)_{te}^{1.03} \qquad (2)$$

For purposes of comparison with the counterflow results presented in reference 3, it was necessary to determine the Reynolds number for flow in the leading-edge passage based on the spanwise length b from the blade tip end of the passage to the thermocouple location and on the average coolant core temperature $(T_5 + T_6)/2$. The Reynolds numbers were calculated from

$$Re_{x,le} = Re_{d,te,f} \frac{\mu_f}{\mu_{av}} \left(\frac{d_{te}}{d_{le}}\right)^2 \frac{b}{d_{te}} \qquad (3)$$

It was also necessary to calculate the Grashof number with the length b.

The results of reference 3 are for air flowing in a vertical heated tube. Reference 4, however, demonstrates the applicability of the correlations of reference 3 when water flows upward in a heated vertical tube. For this reason and the fact that the Prandtl numbers for water and air are not greatly different at the temperatures considered, the use of the counterflow results of reference 3 is justified for the purposes of this report. Values of Reynolds number and the product of Grashof and Prandtl numbers were calculated for the maximum- and minimum-size leading-edge passages. By plotting the values for the 0.50-inch-diameter passage on figure 11 of reference 3, the local Nusselt numbers were found to agree quite well with the mean data line for pure natural-convection experiments. Figure 6 of reference 3 shows good agreement between the mean data line and theory. For the 0.125-inch-diameter passage the Nusselt numbers were no more than 50 percent above the natural-convection data. As mentioned previously, variations of ±50 percent in Nusselt number result in small changes of blade temperature. Therefore, it was considered adequate for blade-temperature calculation to use the theoretical relationship for heat transfer under conditions of free-convection flow over flat plates and in tubes of large diameter (eq. (1)).

Cooling Effectiveness

A more direct means of determining the effect of crossover passages on blade-cooling characteristics is the comparison of the experimentally measured temperatures for blade configurations with and without crossover passages. This comparison is valid if the conditions of operation and the blade geometries are the same. Except for small variations in gas temperature and coolant inlet temperature, such a comparison could be made of the results of this investigation with the results of reference 2. Equation (20) of reference 5 describes the temperature distribution in a body through which heat is conducted from a hot gas to a liquid coolant. From this equation a cooling effectiveness can be defined for the desired comparison:

$$\Phi = \frac{T_{g,e} - T_b}{T_{g,e} - T_{in}} \tag{4}$$

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such that Φ is predominantly a function of the coolant weight flow into the rotor. Since the values of Φ for the conditions of comparison are between 0.5 and 1.0, it was more convenient to plot the quantity $1 - \Phi$ when a logarithmic scale was used.

RESULTS AND DISCUSSION

The object of this report is to determine experimentally the effect of crossover passages in the blade tip on the circulation of coolant through passages in turbine rotor blades. Coolant circulation in small-diameter passages may be improved with crossover passages over that obtained in blind passages (see ref. 1). The adequacy of the size of the leading-edge passage, which serves as the reservoir for coolant flow through the smaller trailing-edge passage, must be determined. Heat-transfer results, comparison of calculated with measured blade temperatures, and cooling-effectiveness parameter $1 - \Phi$ are used to determine the effect of the blade-tip crossover passages.

The operating conditions and pertinent temperature data for the turbine are summarized in table I. Representative data were selected from the table for graphical illustration of trends or agreement with theory. Reference 2 demonstrates the agreement of calculated with measured blade temperatures near a 0.25-inch-diameter coolant passage over a complete operating range. Since the agreement was good, only arbitrary points were selected to show this comparison for the data of this report.

Measured Blade Temperatures

Typical variations of blade, coolant, and effective gas temperature with coolant flow into the rotor are presented in figure 3 for a turbine speed of 8000 rpm. The blade-to-coolant heat flow on a per-blade basis is included. The blade temperatures vary regularly with the coolant flow and follow quite closely the variation of the coolant temperatures T_5 and T_6. The fact that T_5 and T_6 agree closely indicates that the water in the reservoir has mixed thoroughly and that the core of fluid flowing radially outward is at a nearly constant temperature. This conclusion is based on the following: For $T_5 > T_6$ there must be an interference reducing circulation; otherwise, fluid at station 6 would tend to migrate to station 5. Since the fluid entering the rotor is at a lower temperature than T_6, there is a tendency for this fluid to be circulated to station 5. In this case T_5 would be expected to be less than T_6; actually $T_5 \geqslant T_6$ (table I). The heat flows plotted in the figure were calculated from the coolant-flow rate into the rotor and the temperature rise $T_6 - T_{in}$. The heat flow at the lowest coolant flow is much less than at the higher coolant flows. However, the gas-to-blade or the blade-to-coolant temperature difference does not differ greatly at the lowest coolant flows from that at the higher flows. The conclusion is that the calculated heat flows at the lowest coolant flows are in error. This error arises because a portion of the coolant is discharged from the rotor as steam. Calculations to be discussed later were not made where steaming occurred.

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The differences between the temperatures near the coolant passages T_1 and T_3 and the coolant temperature T_5 are of the order 30° to 50° F. Since these temperature differences are small and the metal temperatures are measured 0.067 inch from the passage, it would be expected that the coolant-passage-surface temperatures must be obtained in order to make heat-transfer correlations that are independent of the geometry surrounding the coolant passages.

Coolant-Passage-Surface Temperatures and Heat-Transfer Correlations

Coolant-passage-surface temperatures were calculated by the method previously described in which the heat-conduction process was taken into account. Using the surface temperature as that from which heat is transferred from the blade, Nu and GrPr were calculated. The characteristic dimension l was used in determining Nu and Gr. The relationship of these dimensionless parameters is presented in figure 4. The temperature at the blade tip T_5 was used as the coolant temperature to which heat is transferred, and the blade-metal temperature was the calculated coolant-passage-surface temperature. Included in the figure are lines representing the equations for natural-convection heat transfer in tubes of large diameter or on vertical flat plates and for heat transfer to one-directional flows generated by free-convection forces in tubes of $l/d = 25$.

The data indicate that the heat transfer is comparable to that predicted theoretically. For a given blade-temperature location, however, there is no apparent trend in the variation of Nu with GrPr. Since no conclusion could be reached regarding the agreement of experiment with theory, it was decided to assume a relation between the Nusselt, Grashof, and Prandtl numbers (eq. (1)) in order to define the heat transfer and then to calculate the blade temperatures near the coolant passages in order to compare them with the experimentally measured values. 3952

Calculated Leading-Edge Temperatures

The calculated blade temperatures near the leading-edge coolant passages are compared with the temperatures measured at these points for the 0.125-, 0.25-, and 0.50-inch-diameter coolant passages in figures 5(a), (b), and (c), respectively. For any configuration, a mean line through the temperature values would show the data to scatter about this mean line within $\pm 15^{\circ}$ F of the measured values. The mean line for the 0.25- and 0.50-inch-diameter passages would almost coincide with the line for one-to-one correlation of calculated with measured temperatures. For the 0.125-inch passage the calculated temperatures do not agree with the measured values as well as the values for 0.25- and 0.50-inch-diameter passages. However, the 20° to 30° F deviation of a mean line through the data is sufficiently accurate for most practical purposes.

It is concluded, therefore, that the theoretical equation of reference 1 describing natural-convection flow in tubes of large diameter (small length-diameter ratio) is adequate to predict the high heat-transfer coefficients and small blade-to-coolant temperature differences in passages in which counterflow exists with calculated through-flow Reynolds numbers (eq. (3)) in the range of 5×10^{3} to 200×10^{3}. Because the blade-to-coolant coefficients are very high with respect to the gas-to-blade heat-transfer coefficients, the blade temperature is largely determined by the coolant temperature.

Calculated Trailing-Edge Temperatures

Calculated trailing-edge temperatures are compared in figure 6 with the measured temperatures. The scatter about a mean line through the data would be approximately $\pm 25^{\circ}$ F, and the location of the mean line shows the calculated temperatures tend to be about 25° F high on the average. This could indicate that the coefficients of heat transfer in the trailing-edge are higher than predicted by the theoretical equation for natural-convection heat transfer in large-diameter tubes. Although improved agreement with measured temperatures might result from use of the equation for heat transfer as obtained in one-directional flow generated by free-convection forces, such calculations are not presented, since for most practical purposes the agreement shown is satisfactory. The magnitude of the scatter and the deviation of the calculated from the measured temperatures for the blades of this report are approximately the same as for the blades of reference 2, where there could be no flow of coolant from leading- to trailing-edge passages. Thus, the theoretical relation for heat transfer by natural convection from flat plates and in vertical tubes of large diameter (small l/d) is adequate to describe the heat-transfer coefficients and small temperature differences obtained in the blades of this investigation.

Cooling Effectiveness

A direct determination of the effect of crossover passages on blade-cooling characteristics can be made by comparing the experimentally obtained cooling-effectiveness parameter $1 - \Phi$ of the blades with and without the crossover passages. For this purpose figures 7, 8, and 9 are presented, in which the cooling-effectiveness parameters as determined at stations 1, 2, and 3 for the 0.50- and 0.125-inch quadrants are compared with those obtained from data of the investigation of reference 2, in which the coolant passages were not connected at the blade tip. For a given effective gas temperature $T_{g,e}$ and coolant inlet temperature T_{in}, it is apparent from the parameter $1 - \Phi$ (see eq. (4)) that progressively lower blade temperatures T_b will result in progressively lower values of the quantity $1 - \Phi$; hence, the lowest value of $1 - \Phi$ represents the greatest cooling effectiveness. The only general trend in effectiveness that is common to the two systems is the increased effectiveness as the coolant flow into the rotor is increased, which corresponds to the decreased blade coolant-passage temperatures T_5 as can be seen from table I.

Values of $1 - \Phi_2$ for the midchord temperature at station 2 are presented in figure 7. For the blades with the 0.50-inch leading-edge coolant passages the maximum effectiveness of the reference 2 investigation, denoted by the lower dashed line, corresponds approximately to the mean effectiveness of the blades of this investigation over the common coolant-flow range (fig. 7(a)). At the conditions of operation of 800° F effective gas temperature and 80° F coolant inlet temperature, the mean effectiveness for the blades with crossover passages would result in a temperature about 10° F lower than that resulting from the mean effectiveness of the blades without crossover passages. Similar results are obtained for the blades with the 0.125-inch leading-edge coolant passages (fig. 7(b)). The conclusion based on the data shown for blade-temperature station 2 is that, while the addition of crossover passages at the blade tip resulted in a slight improvement of cooling

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effectiveness, the actual magnitude of the blade-temperature reduction would be of little concern for most practical applications in the range of conditions investigated.

Similar conclusions are obvious when cooling-effectiveness comparisons are made for the blades with and without crossover passages at temperature stations 1 and 3, as shown in figures 8 and 9, respectively. The effectiveness of the trailing-edge passage associated with the 0.50-inch leading-edge passage is improved more than in the trailing-edge passage associated with the 0.125-inch leading-edge passage when blades with and without crossover passages are compared. Thus, the 0.50-inch 3952
passage is a more adequate reservoir for coolant supply to a 0.10-inch-diameter passage than is a 0.125-inch-diameter passage. Again, computing the actual blade temperatures from the mean-effectiveness values based on figure 9(a) discloses that the temperature of the blade with a crossover passage is at most 25° F lower than that of the blade without the crossover passage.

The evidence presented herein indicates a slight improvement in cooling effectiveness of blades with crossover passages over that for blades without crossover passages. In particular, the temperature near a 0.10-inch-diameter passage ($l/d = 25.5$) is reduced more when the coolant reservoir is a passage of $l/d = 5.1$ than when the reservoir is a passage of $l/d = 20.4$. The provision of blade-tip crossover passages for loop-type circulation results in blade temperatures lower than when the crossover passage is not provided, limiting the circulation to pure free convection. However, the temperature reduction involved is probably slight for most practical applications. In the realm of high blade-to-coolant heat-transfer coefficients as obtained with water, the blade temperature is largely determined by the coolant temperature.

SUMMARY OF RESULTS

The results of an investigation of the characteristics of water-cooled turbine blades in which the coolant passages are connected by crossover passages at the blade tips are as follows:

1. The high heat-transfer coefficients and small temperature difference between blade coolant-passage surface and cooling water obtained in the turbine of this investigation can be adequately described by natural-convection theory for heat transfer from flat plates or large-diameter tubes. Calculated blade-metal temperatures including a metal heat-conduction correction agree satisfactorily with measured values. Blade temperatures are determined largely by the coolant temperature when the blade-to-coolant coefficients are high as obtained with water as the coolant.

2. The magnitude of temperature reduction obtained for passages of $l/d = 25.5$ when loop-type circulation is used is probably slight for most practical applications regardless of the coolant reservoir (passages of $l/d = 5.1$ to 20.4).

3. Passages in the range of $l/d = 5.1$ to 25.5 result in slightly higher cooling effectiveness when crossover passages are provided than when closed-end passages are used. The temperature reductions associated with the improved effectiveness are small, about 25° F maximum, in the range of the operating conditions of this investigation.

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Lewis Flight Propulsion Laboratory
National Advisory Committee for Aeronautics
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TABLE I. - TURBINE OPERATING DATA

(a) Leading-edge coolant-passage diameter, 0.125 inch

Turbine speed, rpm	Effective gas temperature, °F	Combustion gas flow, lb/sec per blade	Coolant flow, lb/sec per blade	Temperature[a], °F: Near leading edge 1	Mid-chord 2	Near trailing edge 3	Trailing edge 4	Coolant inlet	Coolant at tip 5	Coolant outlet 6	Outer rim (b)	Inner rim (b)
4,000	621	0.106	0.013	215	225	210	235	78	130	120		
	628	.106	.011	225	230	220	250	78	135	135		
	643	.106	.0052	265	275	260	290	78	185	180		
	621	.106	.0021	295	305	265	315	81	235	205		
	837	.106	.013	263	297	265	323	88	160	147		
	838	.107	.011	280	312	279	340	78	161	152		
	840	.107	.0053	320	351	323	360	90	227	222		
	836	.107	.0021	325	360	340	372	94	255	223		
6,000	840	0.105	0.013	260	285	245	300	88	160	145		
	821	.106	.011	285	293	261	317	90	168	162		
	834	.107	.0054	325	335	307	355	92	217	215		
	832	.106	.0021	335	350	285	370	96	250	225		
8,000	824	0.106	0.013	260	277	247	300	90	160	150		
	827	.105	.011	270	290	255	307	90	170	162		
	827	.106	.0054	300	325	287	340	93	217	215		
	827	.105	.0022	323	342	288	354	96	242	218		
10,000	845	0.107	0.013	280	302	260	345	80	161	150		
	866	.106	.011	293	313	272	354	82	173	165		
	855	.107	.0053	327	349	280	387	84	223	212		
	843	.107	.0022	343	363	300	398	89	248	223		
	1012	.104	.013	315	350	300	395	82	190	185		
	1005	.105	.011	330	360	315	405	83	210	205		
	1022	.104	.0054	355	390	310	435	86	250	220		
	1068	.107	.0022	397	414	338	470	92	265	227		
	1190	.104	.013	345	385	330	445	83	210	205		
	1205	.105	.011	365	400	395	460	85	230	230		
	1210	.104	.0054	390	420	350	480	90	255	240		
12,000	830	0.106	0.013	295	310	255	340	92	185	195		

[a]Numbered thermocouples refer to thermocouples of fig. 2; others to thermocouples of fig. 1.

[b]No thermocouple here in this quadrant.

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TABLE I. - Continued. TURBINE OPERATING DATA

(b) Leading-edge coolant-passage diameter, 0.250 inch

Turbine speed, rpm	Effective gas temperature, °F	Combustion gas flow, lb/sec per blade	Coolant flow, lb/sec per blade	Temperature[a], °F								
				Near leading edge 1	Midchord 2	Near trailing edge 3	Trailing edge (b) 4	Coolant inlet	Coolant at tip 5	Coolant outlet 6	Outer rim	Inner rim
4,000	621	0.106	0.013	180	190	195		78	135	120	165	135
	628	.106	.011	190	200	205		78	140	135	175	140
	643	.106	.0052	235	250	250		78	195	190	225	200
	621	.106	.0021	275	285	285		81	235	210	260	235
	837	.106	.013	230	250	250		88	152	145	215	150
	838	.107	.011	233	257	255		78	156	150	215	160
	840	.107	.0053	275	300	303		90	207	217	270	215
	836	.107	.0021	307	330	324		94	245	225	295	245
6,000	840	0.105	0.013	225	245	235		88	155	150	204	150
	821	.106	.011	248	263	247		90	170	166	218	167
	834	.107	.0054	280	305	290		92	221	222	265	220
	832	.106	.0021	310	330	315		96	250	230	290	225
8,000	824	0.106	0.013	225	250	230		90	155	150	200	150
	827	.105	.011	232	258	240		90	165	163	210	160
	827	.106	.0054	272	300	280		93	220	212	250	212
	827	.105	.0022	293	325	303		96	245	220	275	242
10,000	845	0.107	0.013	238	267	235		80	158	157	208	163
	866	.106	.011	249	277	245		82	173	173	218	180
	855	.107	.0053	295	322	292		84	227	218	266	232
	843	.107	.0022	311	335	305		89	247	222	283	250
	1012	.104	.013	275	310	270		82	185	185	235	185
	1005	.105	.011	290	325	285		83	205	205	255	210
	1022	.104	.0054	325	355	320		86	245	215	290	245
	1068	.107	.0022	350	382	344		92	260	230	311	265
	1190	.104	.013	315	350	305		83	205	205	260	210
	1205	.105	.011	335	370	325		85	230	220	285	230
	1210	.104	.0054	360	400	350		90	260	230	315	255
12,000	830	0.106	0.013	237	295	255		92	192	195	222	192

[a]Numbered thermocouples refer to thermocouples of fig. 2; others to thermocouples of fig. 1.

[b]No thermocouple here in this quadrant.

TABLE I. - Continued. TURBINE OPERATING DATA

(c) Leading-edge coolant-passage diameter, 0.375 inch

Turbine speed, rpm	Effective gas temperature, °F	Combustion gas flow, lb/sec per blade	Coolant flow, lb/sec per blade	Temperature[a], °F								
				Near leading edge 1	Midchord 2	Near trailing edge 3	Trailing edge (b) 4	Coolant inlet	Coolant at tip 5	Coolant outlet 6	Outer rim (b)	Inner rim (b)
4,000	621	0.106	0.013	170	180	190		78	125	120		
	628	.106	.011	180	190	200		78	135	130		
	643	.106	.0052	220	235	245		78	185	185		
	621	.106	.0021	265	275	285		81	230	210		
	837	.106	.013	220	235	255		88	155	143		
	838	.107	.011	216	238	260		78	150	148		
	840	.107	.0053	275	285	300		90	215	202		
	836	.107	.0021	295	310	327		94	240	217		
6,000	840	0.105	0.013	210	230	240		88	150	145		
	821	.106	.011	223	248	255		90	161	167		
	834	.107	.0054	265	280	295		92	213	207		
	832	.106	.0021	290	310	325		96	240	220		
8,000	824	0.106	0.013	212	232	235		90	150	145		
	827	.105	.011	220	240	240		90	160	153		
	827	.106	.0054	262	278	278		93	210	195		
	827	.105	.0022	285	303	307		96	235	210		
10,000	845	0.107	0.013	218	250	255		80	156	155		
	866	.106	.011	230	260	265		82	170	168		
	855	.107	.0053	273	300	306		84	217	212		
	843	.107	.0022	295	322	327		89	242	216		
	1012	.104	.013	250	290	280		82	180	180		
	1005	.105	.011	260	300	295		83	195	195		
	1022	.104	.0054	300	340	330		86	240	210		
	1068	.107	.0022	325	362	372		92	255	216		
	1190	.104	.013	285	325	320		83	200	200		
	1205	.105	.011	305	345	345		85	225	215		
	1210	.104	.0054	335	375	375		90	255	220		
12,000	830	0.106	0.013	240	263	263		92	175	168		

[a]Numbered thermocouples refer to thermocouples of fig. 2; others to thermocouples of fig. 1.

[b]No thermocouple here in this quadrant.

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TABLE I. - Concluded. TURBINE OPERATING DATA

(d) Leading-edge coolant-passage diameter, 0.500 inch

Turbine speed, rpm	Effective gas temperature, °F	Combustion gas flow, lb/sec per blade	Coolant flow, lb/sec per blade	Temperature[a], °F								
				Near leading edge 1	Midchord 2	Near trailing edge 3	Trailing edge 4	Coolant inlet	Coolant at tip 5	Coolant outlet 6	Outer rim (b)	Inner rim (b)
4,000	621	0.106	0.013	165	170	185	245	78	115	115		
	628	.106	.011	175	175	190	250	78	125	125		
	643	.106	.0052	215	225	235	290	78	185	185		
	621	.106	.0021	260	265	280	330	81	230	215		
	837	.106	.013	205	220	232	330	88	145	145		
	838	.107	.011	200	224	243	315	78	147	145		
	840	.107	.0053	260	270	285	365	90	220	215		
	836	.107	.0021	285	295	310	371	94	237	223		
6,000	840	0.105	0.013	195	210	215	310	88	145	145		
	821	.106	.011	207	227	238	305	90	160	165		
	834	.107	.0054	258	265	278	350	92	218	220		
	832	.106	.0021	280	295	305	270	96	235	225		
8,000	824	0.106	0.013	205	212	215	310	90	155	150		
	827	.105	.011	212	222	227	320	90	165	162		
	827	.106	.0054	255	263	267	355	93	210	212		
	827	.105	.0022	277	290	293	370	96	232	225		
10,000	845	0.107	0.013	205	232	233	347	80	147	146		
	866	.106	.011	217	240	242	341	82	161	159		
	855	.107	.0053	256	286	285	403	84	214	210		
	843	.107	.0022	285	312	314	424	89	242	224		
	1012	.104	.013	230	270	265	405	82	180	175		
	1005	.105	.011	250	280	280	415	83	195	190		
	1022	.104	.0054	295	325	320	455	86	240	220		
	1068	.107	.0022	320	350	358	503	92	256	232		
	1190	.104	.013	270	305	300	475	83	200	195		
	1205	.105	.011	295	325	320	490	85	220	230		
	1210	.104	.0054	335	355	360	525	90	255	240		
12,000	830	0.106	0.013	227	242	247	345	92	172	185		

[a]Numbered thermocouples refer to thermocouples of fig. 2; others to thermocouples of fig. 1.

[b]No thermocouple here in this quadrant.

[redacted]

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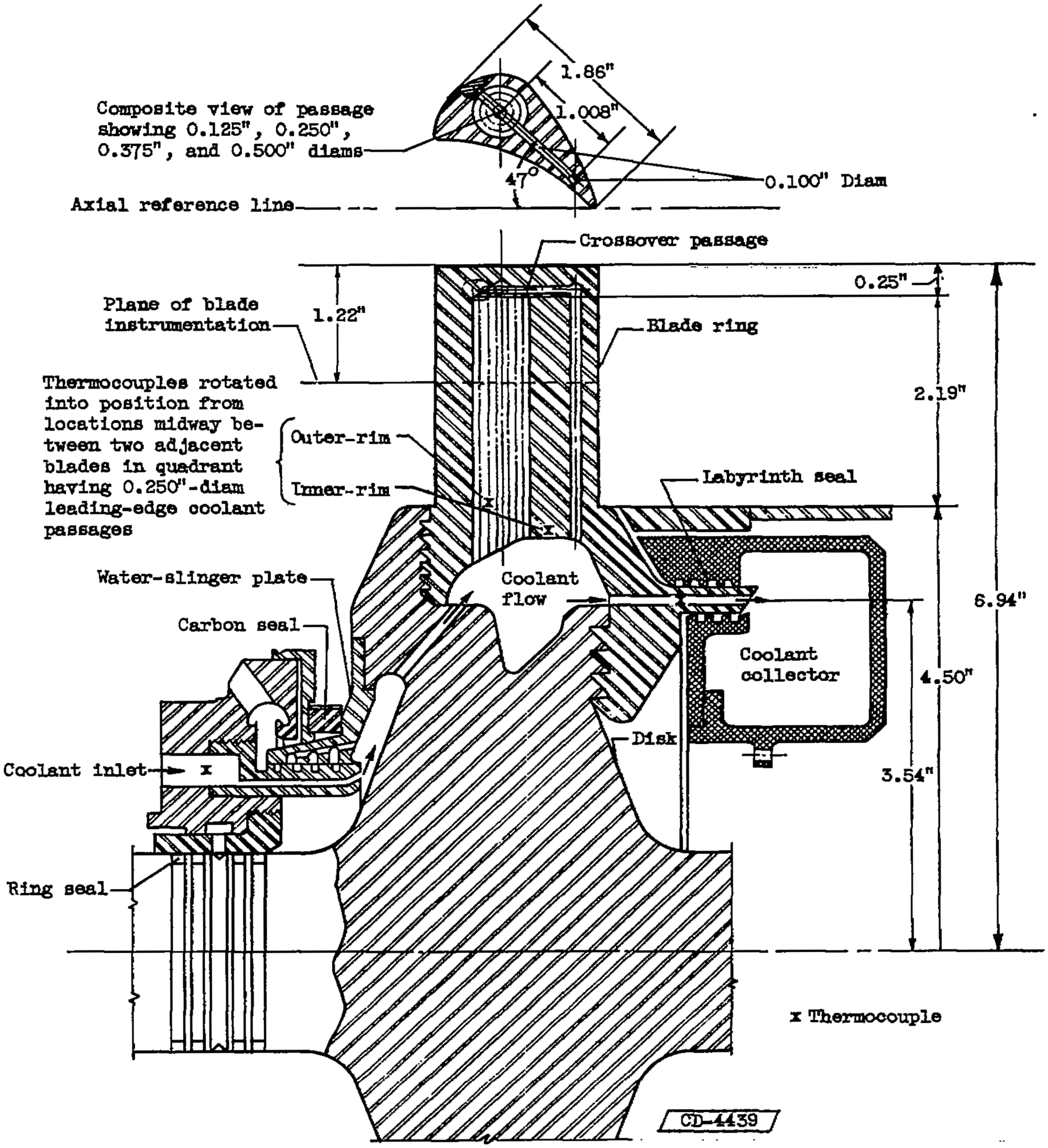

Figure 1. - Cross section of rotor and rotor-coolant system of loop-circuit liquid-cooled stainless-steel turbine showing instrumentation locations.

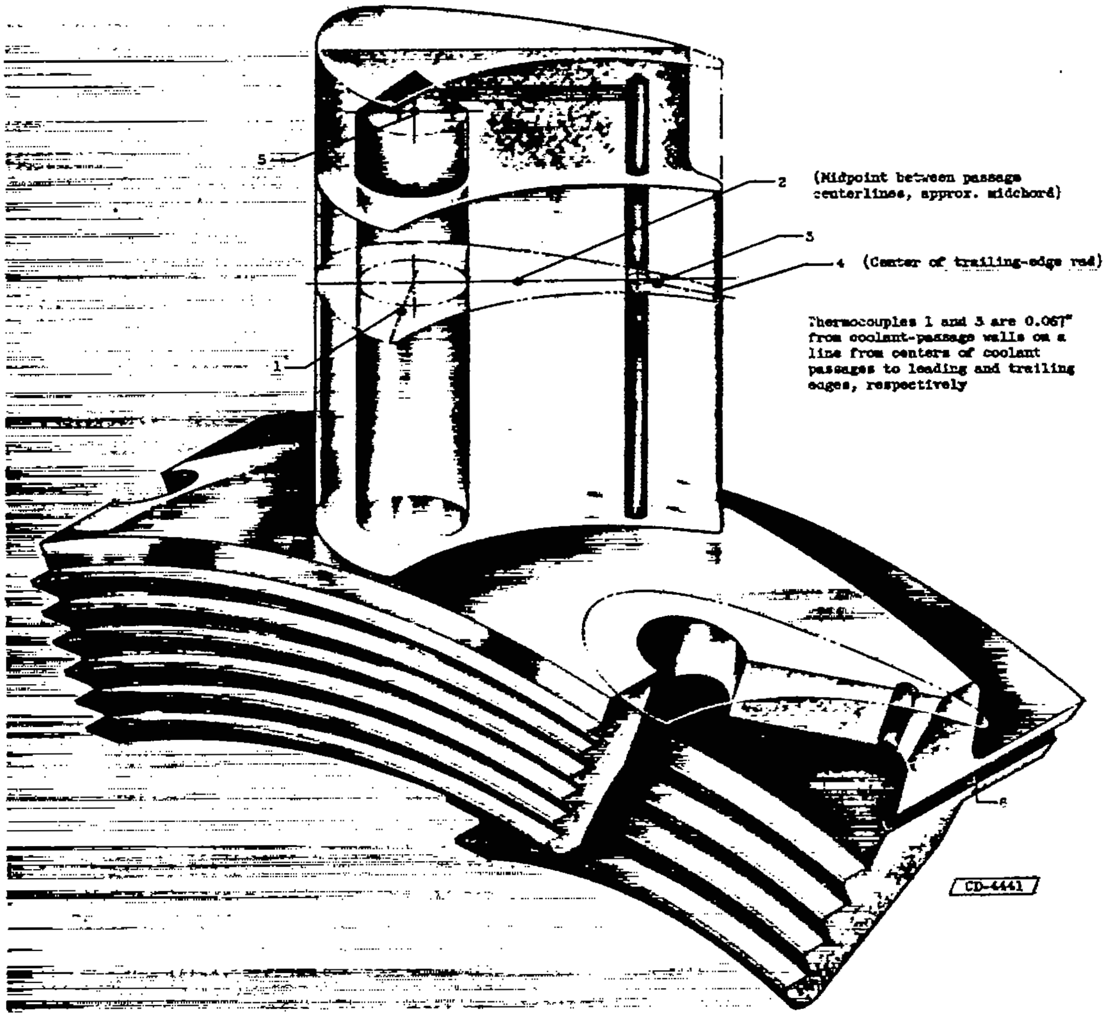

Figure 2. - Diagrammatic representation of rotor blade thermocouple locations on loop-circuit liquid-cooled stainless-steel turbine.

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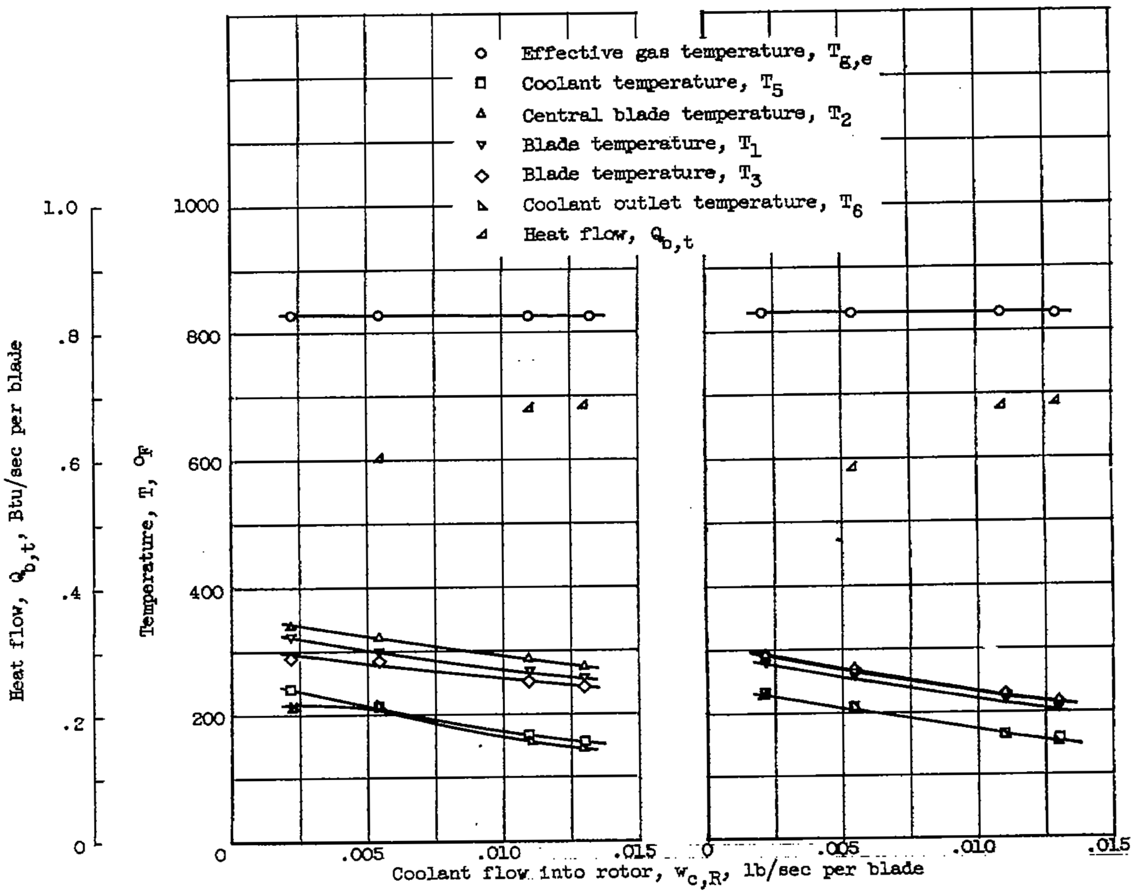

(a) Leading-edge coolant-passage diameter, 0.125 inch.

(b) Leading-edge coolant-passage diameter, 0.50 inch.

Figure 3. - Operating data for water-cooled turbine at rotor speed of 8000 rpm.

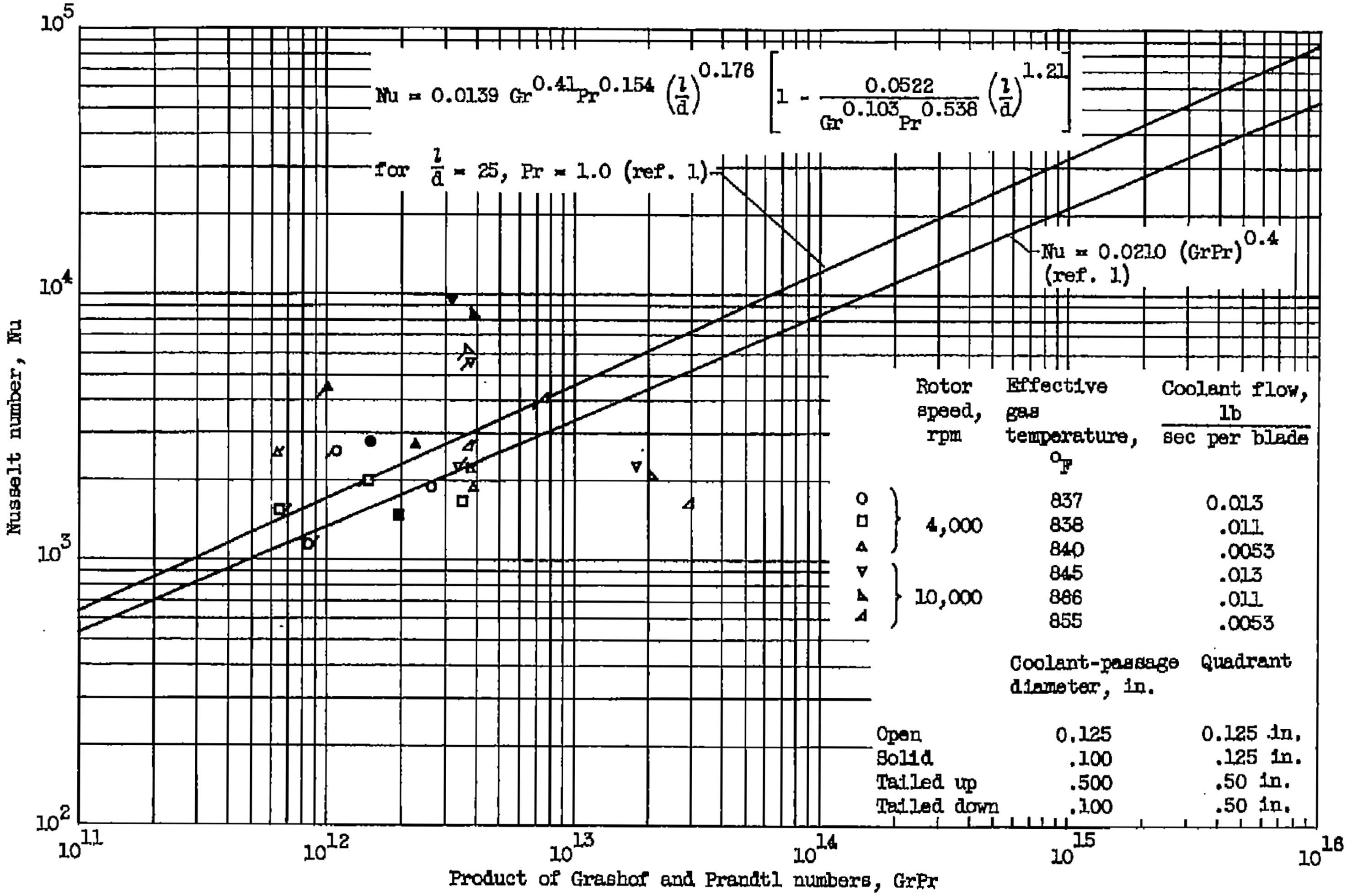

Figure 4. - Dimensionless heat-transfer correlation using calculated coolant-passage-surface temperature.

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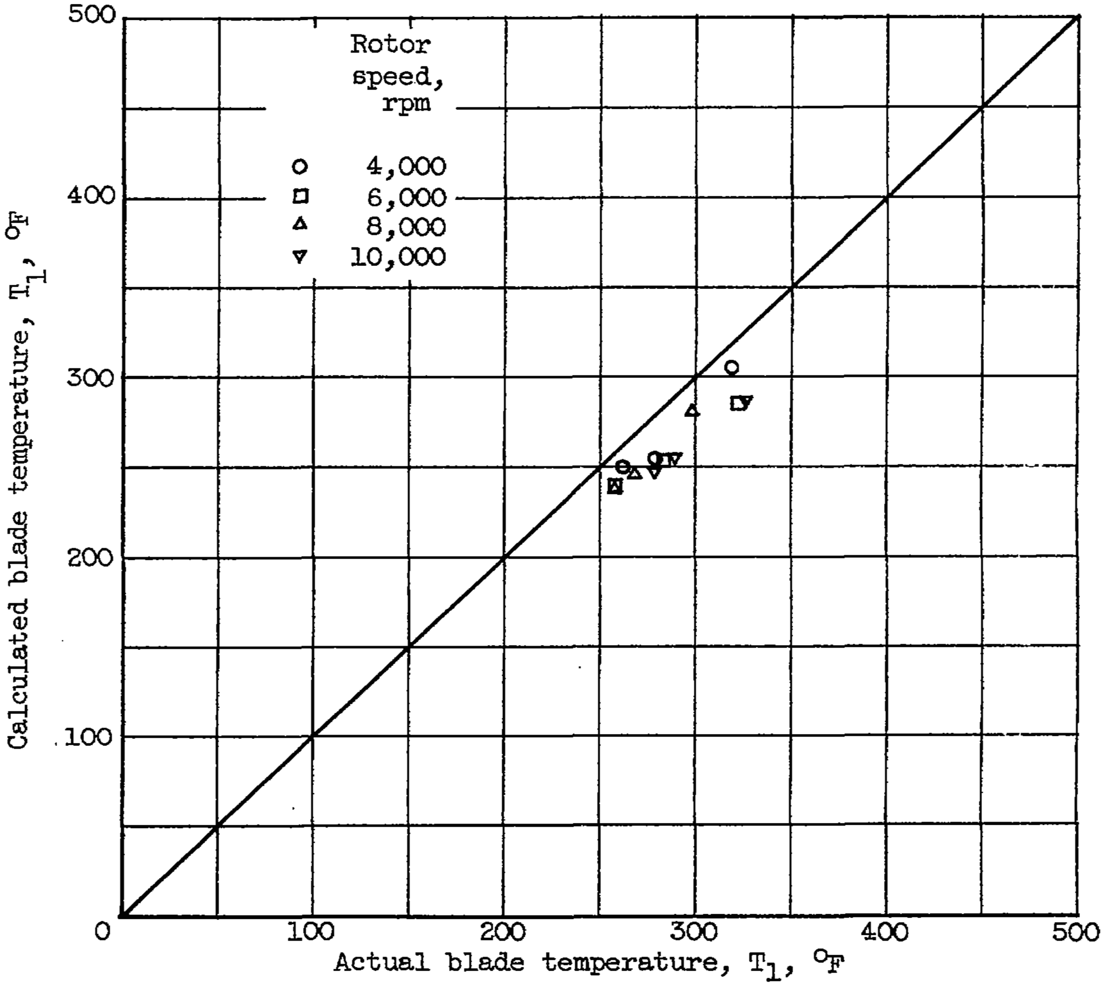

(a) Leading-edge coolant-passage diameter, 0.125 inch.

Figure 5. - Comparison of measured and calculated blade temperatures in leading-edge portion of water-cooled blades.

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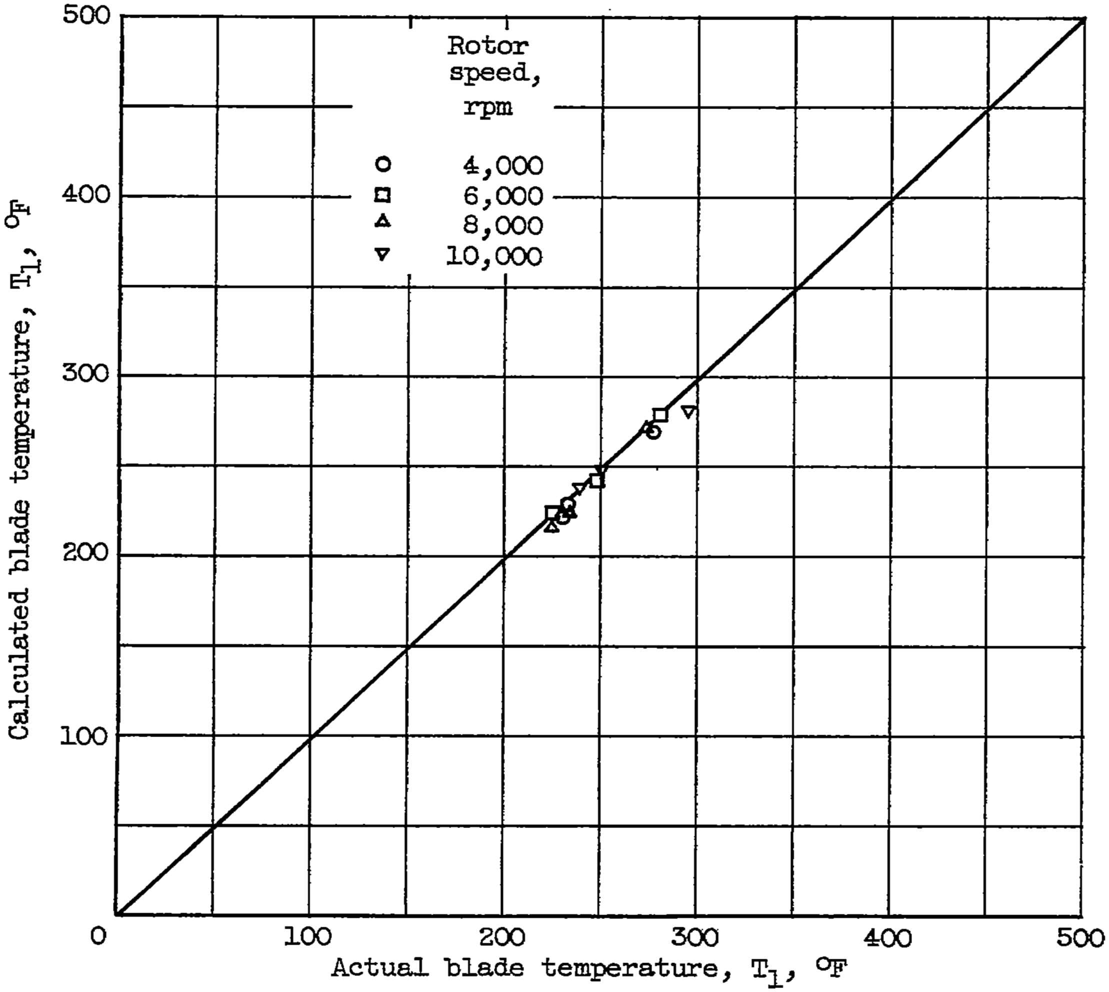

(b) Leading-edge coolant-passage diameter, 0.25 inch.

Figure 5. - Continued. Comparison of measured and calculated blade temperatures in leading-edge portion of water-cooled blades.

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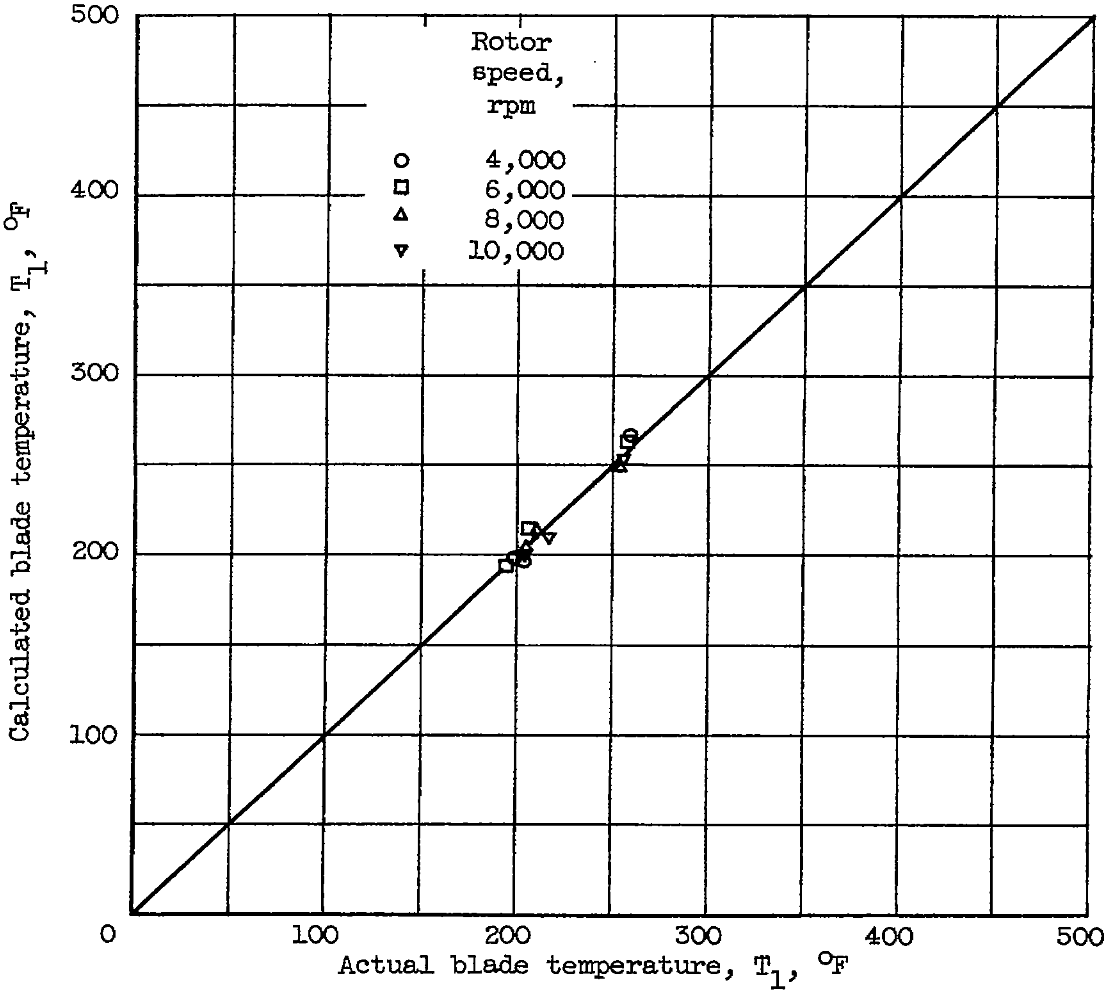


(c) Leading-edge coolant-passage diameter, 0.50 inch.

Figure 5. - Concluded. Comparison of measured and calculated blade temperatures in leading-edge portion of water-cooled blades.

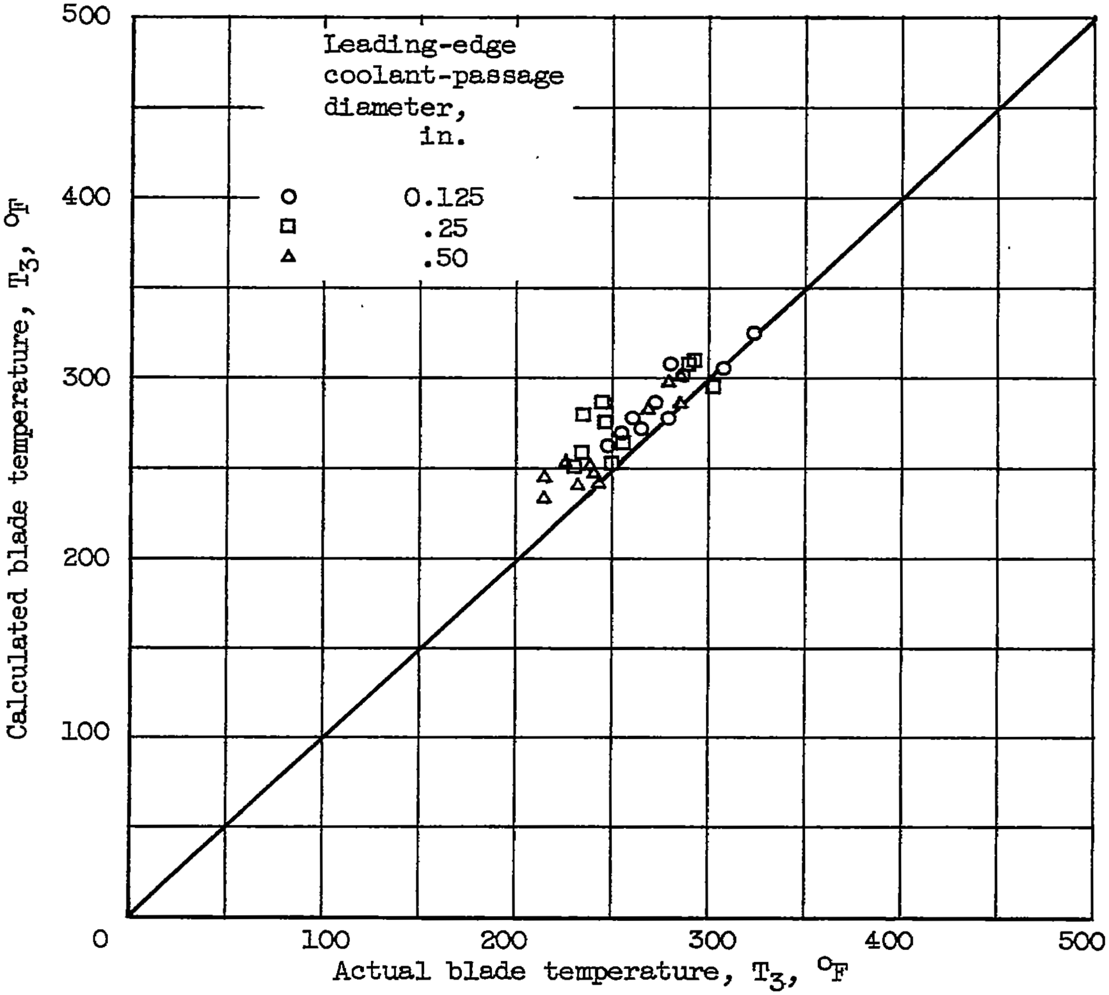
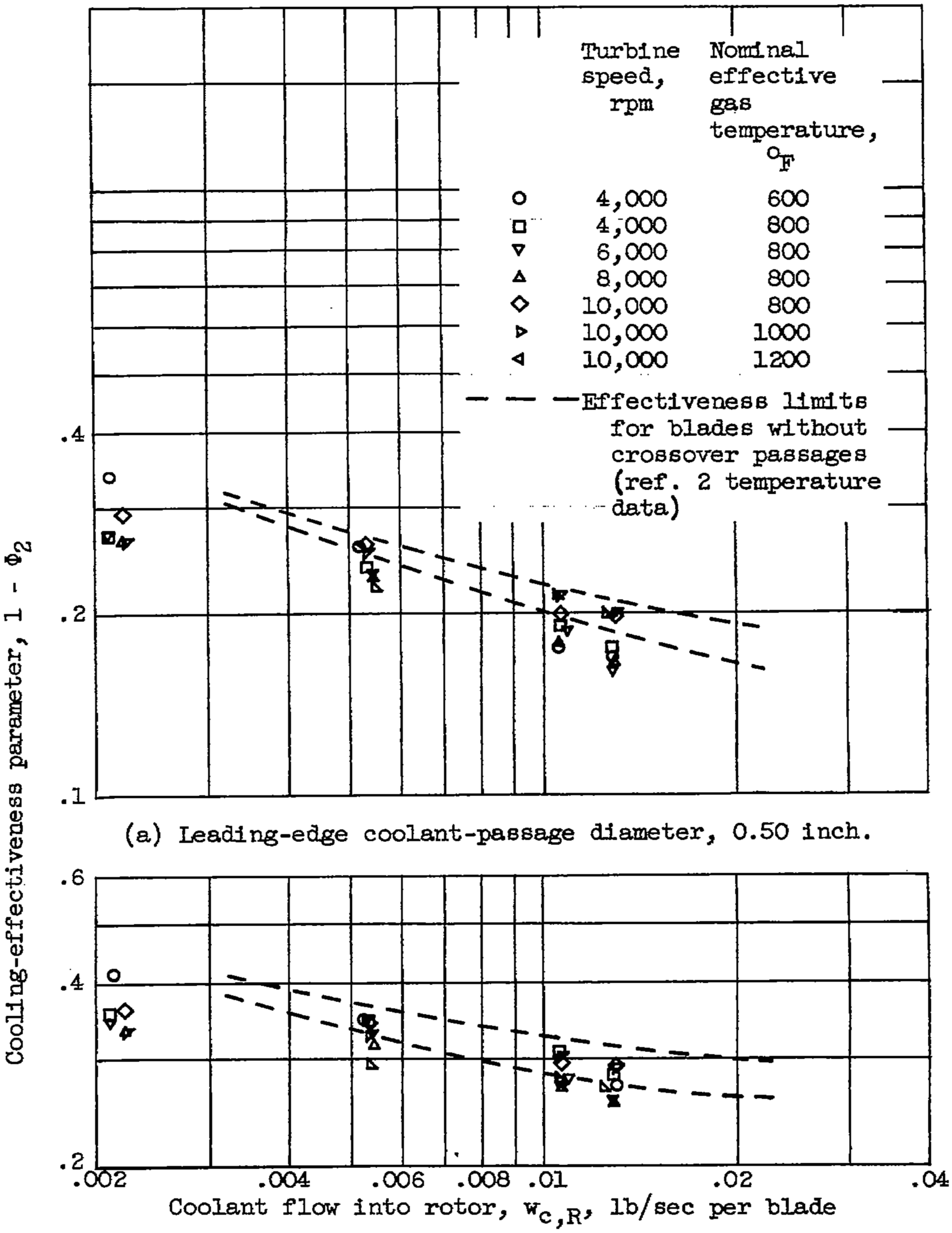


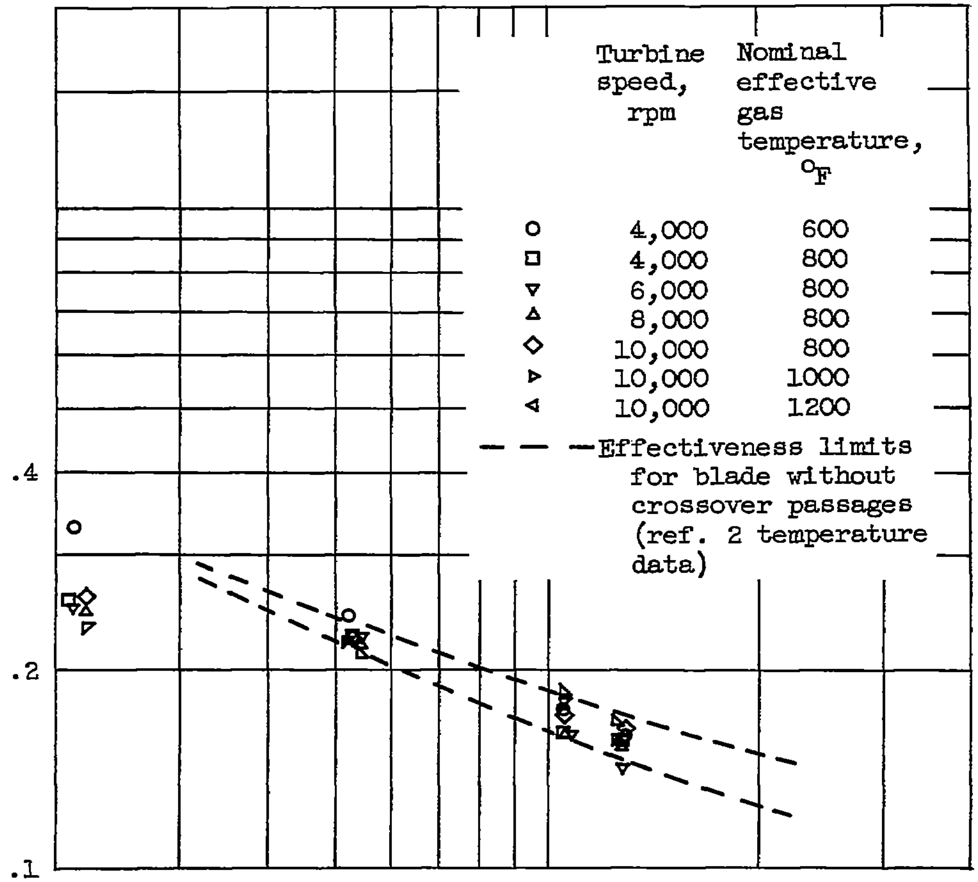
Figure 6. - Comparison of measured and calculated blade temperatures in trailing-edge portion of water-cooled blades.



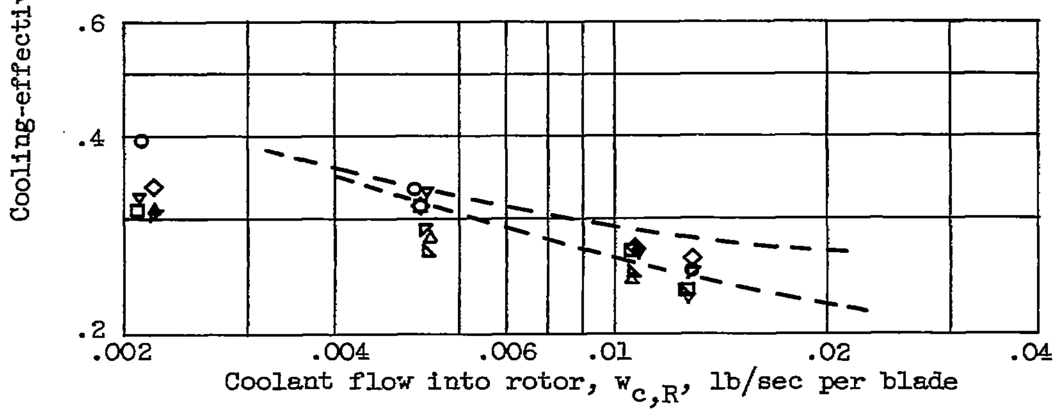
(a) Leading-edge coolant-passage diameter, 0.50 inch.

(b) Leading-edge coolant-passage diameter, 0.125 inch.

Figure 7. - Comparison of midchord cooling effectiveness of blades with and without crossover passages at blade tip.

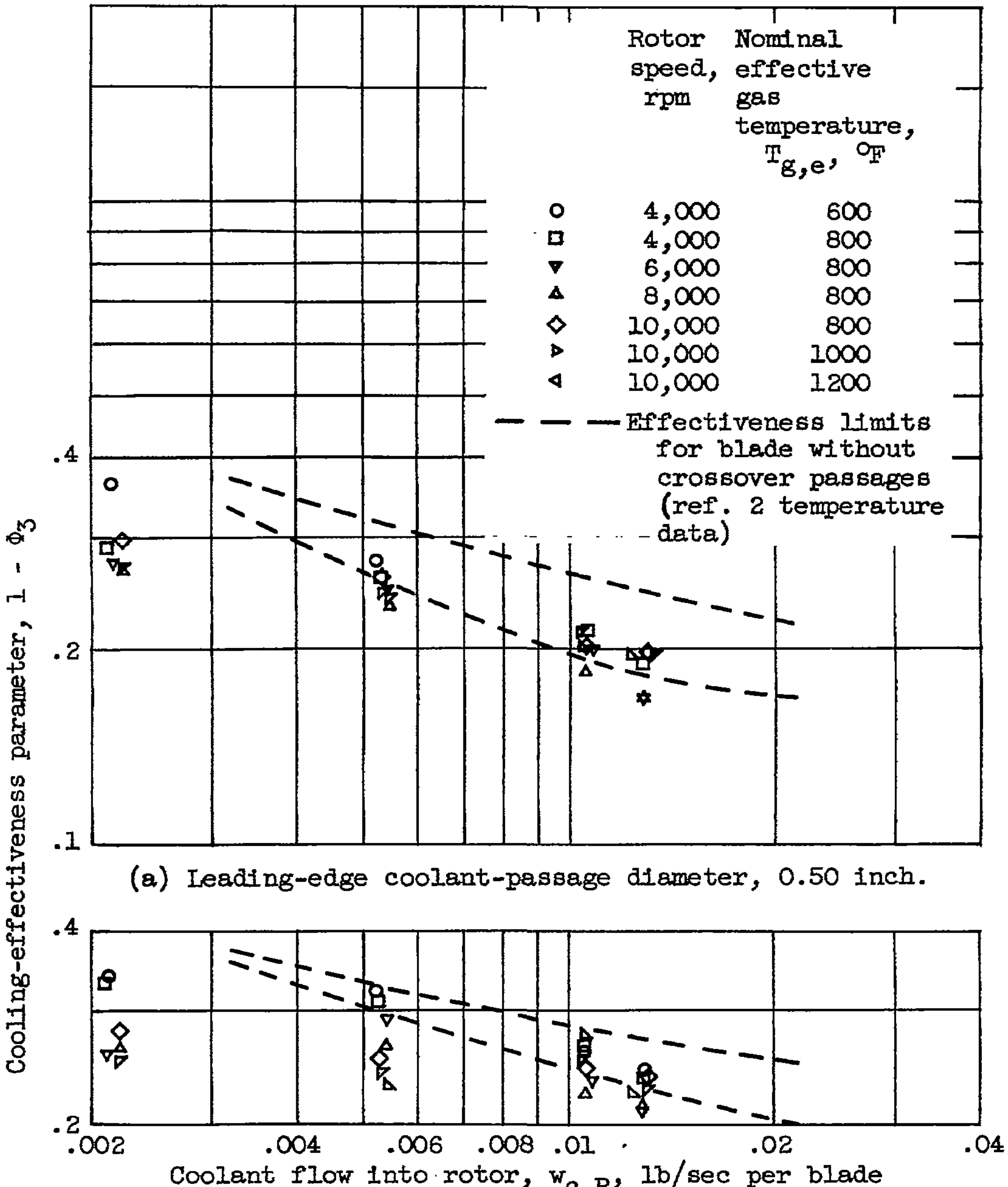


(a) Leading-edge coolant-passage diameter, 0.50 inch.



(b) Leading-edge coolant-passage diameter, 0.125 inch.

Figure 8. - Comparison of cooling effectiveness near leading edge for blades with and without crossover passages at blade tip.



(a) Leading-edge coolant-passage diameter, 0.50 inch.

(b) Leading-edge coolant-passage diameter, 0.125 inch.

Figure 9. - Comparison of cooling effectiveness near trailing edge for blades with and without crossover pasages at blade tip.

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